BASHTA, T.M.

BASHTA, T.M., doktor tekhnicheskikh nauk, professor.

Packing hydraulic units using sectional metal rings. Vest.mash. 37 no.9:20-24 S '57. (MIRA 10:9)
(Packing (Mechanical engineering))

BOGDANOVICH, Leonid Boleslavovich; BASHTA, T.M., prof., doktor tekhn. nauk, retsenzent; LEUTA, V.I., inzh., red.; RUDENSKIY, Ya.V., tekhn. red.

[Hydraulic mechanisms for translational motion; designs and construction] Gidravlicheskie mekhanizmy postupatel'nogo dvizheniia; skhemy i konstruktsii. Kiev, Gos. nauchno-tekhn. izd-vo mashinostroit. lit-ry, 1958. 180 p. (MIRA 11:10)
(Hydraulic machinery)

BASHTA, T.M., doktor tekhn. nauk, prof.

Using rubber rings with circular cross section for sealing hydraulic machinery with straight motion. Stroi. i dor. mashinostr. no.4:9-13 Ap '58. (MIRA 11:4)

(Sealing (Technology))

AUTHOR: Bashta, T.M. SOV/121-58-8-5/29

TITLE: Design and Analysis of Air Loaded Hydraulic Accumulators (Konstruktsii i raschet gidropnevmaticheskikh akkumulyatorov)

PERIODICAL: Stanki I Instrument, 1958, Nr 8, pp 16-18 (USSR)

ABSTRACT: A Variety of designs of piston-type and diaphragm-type hydraulic accumulators with air loading is illustrated and briefly discussed. The well-known analysis to determine the useful and total volumes of the accumulator is recited. Fig 5 reproduces a family of curves for different initial pressures relating the working pressure and the relative oil volume. Fig 6 is a plot of the relative oil volume against the working pressure at different initial pressures in the two cases of

Card 1/2

SOV/121-58-8-5/29

Design and Analysis of Air Loaded Hydraulic Accumulators

iso-thermal and adiabatic processes. A discussion shows that loading the accumulator under iso-thermal conditions introduces a larger percentage of oil volume.

There are 6 figures and 1 reference (German)

Card 2/2

SOV/122-58-8-5/29

AUTHOR: Bashta, T.M., Doctor of Technical Sciences, Professor

TITLE: Contribution to the Problem of the Hydraulic Locking ("Jamming") of Plungers in Hydraulic Units (K voprosu gidravlicheskogo "zashchemleniya" plunzherov gidroagregatov)

PERIODICAL: Vestnik mashinostroyeniya, 1958, Nr 8, pp 18-22 (USSR)

ABSTRACT: In all actual hydraulic systems, the radial distribution of pressure on the plungers is non-uniform as a result of which the plungers do not remain co-axial with the cylinders in which they slide but are pressed to one side of the bore by the unbalanced radial force. Consequently, even if there were no pressure gradient along the passage, there is always a friction force opposing axial displacement of the plunger. This form of sticking in valves and pumps is known as hydraulic locking and is often a source of difficulty to users of hydraulic equipment. Consider the case of a cylindrical piston or valve (shown in Figure 1) of length l with a clearance s between it and the cylinder bore. Let the pressure on the left be p_1 and on the right p_2, where $p_1 > p_2$. To simplify the analysis, the inertia and the gravity forces

Card1/5

of the slider are neglected and the flow of the liquid in the clearance space is assumed to be laminar with a negligible boundary layer. If the axis of the plunger remains parallel to the axis of the bore but is displaced, say, upwards by an amount e so that the upper clearance is $y_1 = s-e$ and the bottom clearance is $y_2 = s+e$ then the clearance will be uniform in the axial direction and therefore the axial pressure drop will be linear, as shown in Figure 1, i.e. there will be a symmetrical distribution of pressure around the plunger. Hence, no out-of-balance, transverse force can arise as all radial forces balance each other.

In the case of a slightly tapered plunger (which is the most likely case in practice due to either an imperfect machining of the parts or as a result of wear either of the plunger or of the cylinder) with the region of higher pressure acting on the base of the frustrum (Figure 2) the pressure gradient is no longer uniform but varies in a parabolic fashion as shown in Figure 2. The larger the amount of tapering the larger the difference in the pressure gradient between the cylindrical and tapered plungers in the axial direction. The total unbalanced force F on the plunger (which is the sum of the vertical components of the pressure only because of the symmetry of the system with respect to the axial vertical plane) acts from the bottom side of the bore upwards. Also, as the average pressure will be lowest on that side of the plunger where it tapers down and highest at the base of the taper there will be an unbalanced moment which will tend to tilt the plunger until it eventually is pressed against the wall of the cylinder, thus producing the hydraulic lock.

If the piston tapers down towards the region of higher pressure (Figure 3) then the maximum gradient of pressure is at the minimum clearance and the piston is pressed to remain in the coaxial position, i.e. the tendency to sticking is smallest in this case.

When a truly cylindrical piston in a truly cylindrical bore is tilted axially, as shown in Figure 4, this represents in effect a gradually increasing clearance on one side and a gradually decreasing clearance on the opposite side of the piston, i.e. the effect is the same as in the case of a tapered piston. In this case, there is a natural tendency to retain the piston in the coaxial position.

The simplest way to equalise the pressure distribution around the piston and to produce a nearly uniform pressure drop in the axial direction is to provide the piston (or the cylinder) with small circumferential grooves as shown in Figure 7 (e.g. 0.3 to 0.5 mm wide and 0.5 to 0.8 mm deep). The sides of the grooves should be perpendicular to the axis of the piston.

Experiments show that even a single groove is very effective; e.g. taking the resistance for an ungrooved

SOV/122-58-8-5/29

Contribution to the Problem of the Hydraulic Locking ("Jamming") of Plungers in Hydraulic Units

piston as 100%, a single groove reduces it to 40% and 7 grooves give only 2.7% resistance against the axial motion of the piston under the same working conditions. There are 8 figures and 3 English references.

1. Hydraulic systems--Performance 2. Hydraulic systems--Equipment

Card 5/5

SOV/84-58-8-30/59

AUTHOR: Bashta, T., Professor, Doctor of Technical Sciences

TITLE: Mechanical and Chemical Stability of Liquids in Aircraft Hydraulic Systems (Mekhanicheskaya i khimicheskaya stoykost' zhidkostey samoletnykh gidrosistem)

PERIODICAL: Grazhdanskaya aviatsiya, 1958, Nr 8, pp 20-21 (USSR)

ABSTRACT: The author discusses the changes taking place in hydraulic liquids under various work conditions, and analyzes the failures caused specifically by the presence in the liquid, in solution or in suspended form of air or water, or both. The solubility of air in oil as a function of pressure is presented in a diagram, and the implications of the phenomenon under various work conditions are analyzed in some detail. Another diagram presents the efficiency drops of a pump depending on the percentage of suspended water in the liquid. Further the causes and effects of emulsification and oxidation of the liquid are discussed. On the whole, the article explains a number of failures most frequently occurring in hydraulic systems. Two diagrams accompany the text.

Card 1/1

BASHTA, T.M.

Face-type packing. Stan. i instr. 29 no.7:29-31 Jl '58.
(Packing (Mechanical engineering)) (MIRA 11:9)

BASHTA, T.M., doktor tekhn. nauk prof.

Design of seals for rotary-type hydraulic machine units. Vest. mash. 38 no.3:24-27 Mr '58. (MIRA 11:2)

(Sealing (Technology))

SOV/86-58-8-19/37

AUTHOR: Bashta, T.M., Professor, Doctor of Technical Sciences

TITLE: Hydraulic Control Servosystems (Gidrousiliteli sistem upravleniya)

PERIODICAL: Vestnik vozdushnogo flota, 1958, 41, No. 8, pp 49-55 (USSR)

ABSTRACT: This informative article deals chiefly with the sensitivity and stability of the system, and ways to increase stability. In this connection, the pertinent points in the operation of the ram and valve assembly are examined in some detail. Insofar as they concern sensitivity, the causes of errors, and methods to minimize them, especially the porting action of the valve spool, and the overlapping of ports are discussed. The relevant kinematics and mechanical arrangements and the range of nonsensitivity are also dealt with. Advantages and disadvantages of "negative overlapping" are given. Plays and clearances, re-

Card 1/2

silience of parts and fastenings, and presence of undissolved air are discussed in their relation to stability. With reference to methods to raise stability, the slotted ports and the damper are described. The need for removal of air, found as an emulsion in the hydraulic liquid, and the need for filtering and watching the effects of low and high temperatures on the liquid are pointed out. The negative and positive aspects of leakage are briefly discussed. The author states that when used correctly, the hydraulic servosystems are reliable. The use of the hydraulic control servosystems improves not only flying and tactical qualities but also extends the life of aircraft. Five figures.

APPROVED FOR RELEASE: 06/06/2000 CIA-RDP86-00513R000203820019-6"

SOV/147-59-2-10/20

AUTHORS: Bashta, T.M. and Nikitin, G.A.

TITLE: Investigation of the Hydraulic Lock Phenomena in Piston Valves (Issledovaniye yavleniya gidrozashchemleniya plunzhernykh par)

PERIODICAL: Izvestiya vysshikh uchebnykh zavedeniy, Aviatsionnaya tekhnika, 1959, Nr 2, pp 83-94 (USSR)

ABSTRACT: This is a work based mainly on the results of references 1-4 as applied to the piston control valves used in aircraft hydraulic systems, consisting of a double ended piston and a cylinder with ports as shown in Fig 1. The clearance between the piston and the cylinder can be made extremely small so that the leakage past the piston lands is very small even at high pressures. On the other hand, it may not be reduced to nothing as this would induce high friction forces. Based on practical experience it seems appropriate to allow about one micron for each 2.5 mm of the piston diameter. Due to pressure difference at the two ends of the valve, there is a flow of liquid from the region of the higher pressure

Card 1/6

towards the region of the lower pressure. For viscous flows in a narrow clearance the rate of flow is given by Eq (1) whose solutions for the three different types (I, II and III) of the clearance as shown in Fig 2 are given by Eq (2), (3) and (4) respectively. Fig 3 shows the axial pressure drop along these clearances. For the parallel clearance the pressure drop is linear, i.e. depends only on the axial position of the given point. But for the divergent and convergent clearances it is curvilinear and depends not on the axial distance (x) but also on the size of the inlet clearance (h_1) and the tapering ratio (m). Figures 4, 5, 6 and 7 show the effect of these as follows: Figures 4 and 5 apply to the divergent clearance and Figures 6 and 7 apply to the convergent clearance, in the first diagram, in each case, h_1 being constant and the tapering increases, while in the second diagram the tapering is constant but the initial clearance decreases. Due to the asymmetrical distribution of pressure around the piston there appears a transverse force N which displaces the piston further and further until it eventually contacts the cylinder, thus evoking a frictional force which opposes any axial movement of the piston. As the coefficient of viscous friction decreases with the thickness of the oil film (see Fig 8 and Ref 5) it is seen that the locking force increases not only because the transverse force on the piston increases with the pressure in the cylinder but also because the coefficient of friction increases: at first on account of a decreased clearance gap (and hence decreased the oil film) and then on account of the viscous friction being transformed into semi-dry friction when the piston is in contact with one side of the cylinder. Fig 9 shows under what conditions a locking force can be produced: I - the clearance is parallel, the pressure distribution is axisymmetric and, therefore, there is no transverse force on the piston; II - unstable position of the piston with a sideways thrust on the piston tending to push it back into its

Investigation of the Hydraulic Lock Phenomena in Piston Valves

central position; III - divergent clearance, piston axis displaced to one side by an amount ε due to which the pressure distribution is asymmetrical and a transverse locking force is directly proportional to ε; IV - Converging clearance: when piston is displaced sidewards there will be a restoring transverse force produced, which pushes the piston back into its central position, i.e. the piston is in a stable equilibrium when situated centrally in its bore; V - local attachment of some foreign body also produces a transverse locking force N. The magnitude of the locking force for the case of divergent clearance is found from Eq (6) by integration, the solution being Eq (7) which is obtained on the assumption that the flow is laminar, entirely in the axial direction and the coefficient of friction being constant. The last relation may be expressed in a dimensionless form as given by Eq (8), by dividing it by a force $N = p_0 2 \ell_0 r$, which is a reference force, and by introducing the relative taper ratio $k = \tau/c$ (see Ref 3). This is called the normalised pressure. Fig 10 shows its dependence on the taper ratio when $\varepsilon = 0$. Experiments confirm these theoretical formulae as shown in Fig 11 where the effect of pressure on the locking force is given for the pistons A and B shown in Fig 12.I and the table under the figure (for the piston of Fig 12.II there is no locking force). A simple method to reduce the unbalanced radial pressures on the piston is to make small circumferential grooves in it (Fig 13 and 14). Experiments with one type of valve showed that a single groove reduces friction force from 100 to 40% and seven grooves give only as little as 2.7%. Since grooves and tapering of the piston help to maintain the piston in its central position, this means that they reduce leakages as well, because leakage increases directly proportional to the eccentricity of the piston position in the cylinder. To reduce the friction more radically, the piston may be made to revolve or to oscillate about its axis. This increases the film thickness between the rubbing surfaces as

SOV/147-59-2-10/20

Investigation of the Hydraulic Lock Phenomena in Piston Valves

shown in Fig 15 and substantially reduces friction. Eg for a 55 mm piston with a flow of 120 l./min of the liquid under a pressure of 6 kg/cm^2, the friction force of an ordinary piston valve was found to be 5 kg, while the same valve with rotating cylinder suffered only 70 gr frictional force. There are 15 figures and 6 references, 2 of which are Soviet and 4 English.

ASSOCIATION: Kiyevskiy aviatsionnyy institut GVF, Kafedra gidravliki (G.V.F.Institute of Aeronautics of Kiyev, Chair of Hydraulics)

SUBMITTED: December 7, 1958

Card 6/6

BASHTA, T.M., doktor tekhn.nauk, prof., laureat Stalinskoy premii

Progress of civil aviation. Nauka i zhyttia 9 no.1:23-26
Ja '59. (MIRA 12:1)

(Airplanes)

BASHTA, T.M.

report to be presented at the 1st Intl Congress of the Intl Federation of Automatic Control, 25 Jun-5 Jul 1960, Moscow, USSR.

AGEYKIN, D. I. - "Compensating thermo-magnetic gas analysers"
ANDREYEV, N. I. - "Method of determining the optimum dynamic system according to the criterion of the functional extra, which is a given function of several other functions"
AYZERMAN, M. A., and GANTMAKHER, F. R. - "Some problems of the theory of nonlinear systems of automatic regulation with discontinuous characteristics"
BARBASHIN, Ye. A. - "Concerning the organization of the LYAPUNOV function for nonlinear systems"
BASHARIN, A. V. - "Graphic methods of synthesis of nonlinear systems of automatic regulation"
BASHTA, T. M. - "Problems of the application of high liquid pressures for automatic systems"
BEDEL'BAYEV, A. K. - "The theory of stability of regulation systems"
BELIKOV, Yu. A. - "Multicoordinate nonlinear interpolator for program control of machines"
BERENDS, T. K., and TAL', A. A. - "Pneumatic alloy systems"
BERNSHTEYN, V. L., KALASHNIKOV, V. I., KNYAZHEVSKIY, V. V., MAKIN, L. V., POPOV, G. A. - "Automated electric drive of the propeller installation of the atomic icebreaker 'Lenin'"
BESEKERSKIY, V.A., and FEDOROV, S. M. - "Application of the equivalent transmission function in the calculation of follower systems by the logarithmic frequency curve method"
BILIK, R. V., MIKELADZE? V. A., and PRANGISHVILI, I. V. - "Contactless telemechanical systems with temporary separation of channels"
BOLTYANSKIY, V.G., GAMKRELIDZE, R. V., MISHCHENKO, E. F., and PONTRYAGIN, L. S. - "The maximum principle in the theory of optimum control processes"
BROZGOL', M. M. - "Automated electric drives of a metallurgical plant"
BUROVOY, I. A. - "Automatic regulation of froth-layer processes in nonferrous metallurgy"

PHASE I BOOK EXPLOITATION SOV/4615

Bashta, Trifon Maksimovich, Professor

Gidravlicheskiye sledyashchiye privody (Hydraulic Servodrives) Moscow, Mashgiz, 1960. 281 p. 14,000 copies printed.

Reviewer: V.A. Leshchenko, Candidate of Technical Sciences: Eds.: A.I. Belevitin, and V.V. Mayevskiy; Chief Ed. (Southern Department, Mashgiz): V.K. Serdyuk, Engineer.

PURPOSE: This book is intended for technical personnel concerned with hydraulic servodrives and their operation.

COVERAGE: The book describes hydraulic servodrives for position control and servomechanisms employed in control systems of various machines and presents information on their manufacture. General and hydraulic calculations of servodrives and their elements are given. Problems of sensitivity and stability of hydraulic servodrives are discussed in detail and various factors affecting them are analyzed. No personalities are mentioned. There are 20 references: 10 Soviet, and 10 English.

Card ~~1/~~6

S/122/60/000/001/001/018
A161/A130

AUTHOR: Bashta, T. M., Doctor of Technical Sciences, Professor

TITLE: The aspects of fluid application in hydraulic machine systems

PERIODICAL: Vestnik mashinostroyeniya, no. 1, 1960, 3-10

TEXT: The article is a general review in digest form, dealing with mineral oils being used for the majority of modern hydraulic machine systems, their properties, peculiarities and stability in service. Mineral oils can lose the viscosity in service, oxidize in contact with air and water or dirty (paint, dust, metal and rust particles). The Stokes law is used in calculation of the settling speed of particles in oils. Formulae are included: of gas solubility in liquids (in connection with its separation and bubbles, and detrimental effect of air accumulations in systems); the variations of the fluid elasticity module or compressibility with varying content of nondissolved air; compression of oil-air mixture, i.e., the hydraulic pump capacity loss. Volume losses through the compression of air can be high at certain relations of the dead space in a pump to the work space. In the example of 5% nondissolved air and dead space/work space relation = 0.1, at nominal work pressure 200 kg/cm^2, atmospheric pressure

Card 1/2

= 1 kg/cm^2 and pressure in the pump suction chamber = 0.5 kg/cm^2 - the volume efficiency of the pump is calculated to be = 0.89. The air content of the pump efficiency varied and pressure is taken from a hydraulic pump chamber which was not completely filled with fluid - pressure shocks reached 380 kg/cm^2 (at nominal work pressure 210 kg/cm^2). Shocks had been observed exceeding 2.5 - 3 times the work pressure in the system. There are 5 figures and 4 non-Soviet-bloc references. The references to the English-language publications read as follows: Ref. 1: "Aircraft Engineering", no. 273, 1951; Ref. 2: "Aircraft Engineering", May 1952; Ref. 3: "ASME", v. 66, 1944; Ref. 4: "SAE", 4/XI, 1943.

69312

26.4000

S/147/60/000/01/001/018
E022/E535

AUTHORS: Bashta, T. M. and Nikitin, G.A.

TITLE: ~~Investigation of~~ Friction Force in Hydraulic 3 Piston-Valve Units

PERIODICAL: Izvestiya vysshikh uchebnykh zavedeniy, Aviatsionnaya tekhnika, 1960, Nr 1, pp 3-11 (USSR)

ABSTRACT: The article is an extension of earlier work, published in Nr 2, 1959 of this journal, and deals with the friction forces between the piston-valves 3 and their casings, the influence of high pressure on the hydraulic lock as well as the effect of the total fall of input pressure on the friction force. The characteristics of systems with very high working pressure differ from those of systems with low and medium pressures. As shown in the earlier paper, the force required to move the piston-valve increases steadily with pressure but when the working pressure is of the order of hundreds of atmospheres new factors appear and influence the locking of the valve. Both the piston and the cylinder will deform at these pressures

Card 1/8

4

69312

S/147/60/000/01/001/018
E022/E535

Investigation of Friction Force in Hydraulic Piston-Valve Units

and hence the clearance changes so that the geometry and configuration of the system alters. Also the viscosity of the fluid along the clearance at these pressure gradients can no longer be considered as constant. Fig 1 shows the axial and transverse section of a hydraulic valve pair, the inner and outer radii of the cylinder being r_1 and r_2 respectively, while the maximum radius of the valve is r_3 and it tapers down to $(r_3 - \tau)$. The radial deformation of the cylinder due to pressure p is then given by Eq (1), where σ is the Poisson's ratio for the cylinder material. Similarly the radius of the piston will decrease by an amount given in Eq (2), $r_{пл}$ being the local value of the piston radius. Hence the total increase of the clearance is the sum of the two as given by Eq (3), in which k is a constant which depends on the geometry of the system and the properties of the materials used. Eq (3) does not include the effects of the tangential stresses

Card 2/8

69312

S/147/60/000/01/001/018
E022/E535

Investigation of Friction Force in Hydraulic Piston-Valve Units

resulting from the pressure change (gradient) along the cylinder. Since the maximum pressure is at the inlet p_1 and diminishes along the cylinder (to p_2) hence the clearance will change in a similar manner, e.g. for the system as in Fig 1, when $r_1 = r_{пл} = 6$ mm, $r_2 = 12$ mm, $\tau = 0.002$ mm, $\ell_o = 10$ mm, c = 0.005 mm material of the piston and cylinder being the same (cast steel 12KhNZA) with $E = 2.04 \cdot 10^6$ kg/cm^2 and $\sigma = 0.28$, then with exit pressure $p_2 = 1$ kg/cm^2 the change in clearance for various pressures is as quoted in the table on p 5, from which it can be seen that very high pressures produce changes in the clearance of the same order as their initial values or of the order of the tapering. Thus in the case of a system with initially increasing clearance (susceptible to hydraulic locking) deformation produced by the pressure will tend to reduce the tapering, so that the clearance will become uniform at first and then, at very high

Card 3/8

69312

S/147/60/000/01/001/018
E022/E535

Investigation of Friction Force in Hydraulic Piston-Valve Units

pressures, the clearance will actually decrease; under these conditions the valve will not get locked. Experiments confirm this conclusion. In the example considered above, the uniformity of the clearance gap was obtained at 300 kg/cm^2. In the case of small inaccuracies of the shape the piston may become free from locking forces even at relatively low pressures, as shown in Fig 2, where the full line represents the force (F, in g) required to move the piston from rest, and the dotted line represents the electric resistance, in ohms, of the clearance (the lubricant between the two metals acts as a dielectric). The graph in Fig 2 is the result of experiments carried out on a system as shown in Fig 3, where (1) is the manometer registering the inlet pressure, (2) is the valve, (3) is the cylinder, (4) is ohmmeter. The range of pressures used was from 0 to 300 kg/cm^2 by increments of 25 kg/cm^2. The pressures were kept constant for one

Card 4/8

69312

S/147/60/000/01/001/018
E022/E535

Investigation of Friction Force in Hydraulic Piston-Valve Units

minute before the readings were taken. Each experiment was repeated three times. The initial force needed to move the piston with no pressure was 44 g; it increased to 1500 g at 75 kg/cm^2 and then decreased until, at 200 kg/cm^2, the piston was completely free from locking, the force needed for the axial movement becoming again 44 g for all pressures up to 300 kg/cm^2. With no inlet pressure into the lubricant the resistance of the system oscillated between 0.5 to 3 ohms. As the pressure increased the valve was pushed towards the wall on one side squeezing out the lubricant from that side until full contact was established so that resistance fell to zero. This happened at pressure of about 25 kg/cm^2 and conditions remained static up to 150 kg/cm^2, after which the resistance began to increase (2.8 ohms at 200 kg/cm^2 11 ohms at 250 kg/cm^2) indicating that the contact between the piston and the cylinder was broken and the valve became displaced towards the central position. One of

Card 5/8

69312

S/147/60/000/01/001/018
E022/E535

Investigation of Friction Force in Hydraulic Piston-Valve Units

the causes of hydraulic jamming of valves may be the insufficient elasticity of the housings. These, very often, may have a variable outer form so that the thickness of the wall is variable. When the initial clearance is small, it is possible, therefore, that at high pressure with deformed piston the magnitude of the clearance in some places will increase while in the other will decrease so that the piston becomes jammed and no movement is possible at all. Experience shows that the magnitude of the locking force depends on the time during which the pressure acts on the system. This may be explained by the accumulation in the clearance of small foreign particles from the lubricant and also by absorption of the charged ions by the metallic surfaces. If the inlet pressure is then suddenly reduced to zero, the conditions do not return immediately to those of the initial state and the force needed to move the piston may be many times larger than

Card 6/8

that which would have been required if the pressure were not applied at all. In fact, several movements of the pistons are required before the conditions return to the initial state. The plot, Fig 4, shows the effects of time and pressure on the magnitude of the locking force F. The results were obtained during the tests of a system shown in Fig 1 in which the dimensions were as follows: piston dia. 12 mm, piston width 10 mm, diametral clearance 10 microns, outer diameter of the cylinder 50 mm. Both the piston and cylinder were made of U12 carbon tool steel. The surfaces were heat treated until their hardness reached the value of $R_c = 50$ and then they were ground and polished. Each experiment was repeated three times and the data were taken over the first minute after the pressure was removed. The effect of impurities in the lubricant on the locking force F is shown in Figs 6 and 7, the upper curve referring to the non-purified oil AMG-10 and the lower to the same oil but purified

69312

S/147/60/000/01/001/018
E022/E535

Investigation of Friction Force in Hydraulic Piston-Valve Units

by passing it through special paper filters supported on a metal net. When the purified oil was used, the valve and the piston were washed with aviation gasoline B-70 and then blown through by compressed air. The amount of impurities present in the oil may be judged from the photograph, Fig 5 (magnification 170 X); the left figure is a photo of a sample of fully purified oil and the right figure is a photo of the same oil not purified.
There are 7 figures, 1 table and 2 references, 1 of which is Soviet and 1 English.

ASSOCIATION: Kafedra aeromekhaniki i gidravliki, Kiyevskiy institut GVF (Chair of Aerodynamics and Hydraulics, Kiyev GVF Institute)

SUBMITTED: October 3, 1959.

Card 8/8

S/084/60/000/006/019/020
A104/A029

AUTHORS: Bashta, T., Professor and Komarov, A., Graduate Engineer

TITLE: Pipe Fatigue Fractures

PERIODICAL: Grazhdanskaya Aviatsiya, 1960, No. 6, pp. 29 - 30.

TEXT: The authors discuss the cause and effect of pipe fatigue fractures of aircraft hydraulic equipment. These fractures result from alternating loads caused by mechanical or parametric oscillations of pipes, buckling and radial oscillations being the most dangerous. Figure 1 shows the direction of liquid pressure force inside the pipe. In aircraft pipes the critical speed leading to buckling oscillations is 20 m/sec; particularly dangerous are oscillations caused by liquid pressure pulsation. Extensive research was carried out by the Kiyevskiy institut GVF (Kiev GVF Institute) on circular and elliptic section pipes. Photograph 2 shows fatigue fractures on a 10 x 12 steel pipe caused by buckling and oscillation. Radial oscillations have a high frequency coefficient and are always caused by the pulsation of liquid pressure. In the case of circular section pipes the increased inner pressure causes an even expansion of the pipe diameter

Card 1/2

S/084/60/000/006/019/020
A104/A029

Pipe Fatigue Fractures

and a resonance is only possible if there is a high frequency pressure pulsation source, such as a pump, and repeated reloading can lead to fatigue fractures. Resonance and forced oscillations are particularly dangerous in irregular, i.e., elliptic section pipes, but unfortunately plants have no regulations demanding the rejection of such pipes. Photograph 3 shows a 10 - 12 steel pipe displaying fatigue fractures caused by radial oscillations of the elliptic section. This type of fracture progresses from inside to the surface which makes detection difficult. Tests proved that even a slight irregularity of the pipe section affects its tensile strength and it is suggested that all pipes displaying a section deformation of more than 5% be rejected. Photograph 4 shows fatigue fractures of a ИЛ-12 (Il-12) pipe. Tests were carried out with АМГ-10 (AMG-10) lubricant, nominal operating pressure was 100 kg/cm^2. There are 3 photographs and 1 figure.

Card 2/2

S/122/60/000/009/002/015
A161/A026

26.2182

AUTHOR: Bashta, T.M., Professor, Doctor of Technical Sciences

TITLE: Problems of High Pressure in Hydraulic Systems

PERIODICAL: Vestnik mashinostroyeniya, 1960, No. 9, pp. 14 - 19

TEXT: Hydraulic high-pressure systems are considered in general, i.e., systems used in aircraft, transport machines and others where weight and dimensions are of critical importance. The author points out that in aircraft high pressure gives gain to aircraft weight only to a certain point from which on weight increases again because of heavier walls in hydraulic systems needed to withstand the pressure. The phenomenon of greater accuracy of tracing systems under high pressure is explained by the action of forces in fluids, penetrating into the space between piston and bushing in slide-valve distributer systems and spreading the bushing and compressing the piston ring ends and, at the same time, forcing the piston into accurate concentrical position. The taper in the piston may be one or several microns only, but it completely eliminates jamming. (Experiments were carried out by Engineer G.A. Nikitin in the author's laboratory, Ref. 1). Two designs of seals are described, with rubber rings and leather side

Card 1/5

APPROVED FOR RELEASE: 06/06/2000 CIA-RDP86-00513R000203820019-6"

88353

Problems of High Pressure in Hydraulic Systems

S/122/60/000/009/002/015
A161/A026

gaskets, or consisting of a soft and of a hard-rubber ring, the latter being more reliable. It is known from experience that piston-type pumps can be used only for 280 - 350 kg/cm^2 pressures. A 7-cylinder pump for 280 km/cm^2 and a 5-cylinder one for the same pressure (Figs. 6 and 7) are described, having mushroom piston heads making the pistons turn and spread evenly the oil film on the friction surface and reducing the toppling moment caused by the reaction force of an inclined disc, which is a part of the drive shaft. The pump has a 50 l/min capacity and proved well in a 2,000-h test. The hydraulic engine (Fig. 8) for 350 kg/cm^2, in contrast to the pumps described (with fixed cylinder block), has a rotary cylinder block and plain bearings. The major problem in designing high-pressure systems is the rigidity of hydraulic elements and of the work medium. This problem is discussed and formulas are recommended for the calculation of pumps. It is stressed that in case of superhigh pressures the compressibility of fluids and the deformation of pump chambers has a particularly great effect. A pump with 8-mm piston and 50-mm lift tested with an 850 kg/cm^2 pressure had a volume efficiency of only 46.4%, and the loss through design spaces constituted only 6.3%, while the loss through deformation of pump parts and compression of the fluid made 47.3%. The presence of undissolved air in the fluid is extremely detrimental for the efficiency. It has been revealed that the viscosity of min-

Card 2/5

eral fluids increases almost in geometrical progression with the pressure growing arithmetically. The mineral oil viscosity for the pressure range 0 - 500 kg/cm^2 may be calculated by the formula $E_p = (1 + 0.003\ p)\ E$, where E is the viscosity in Engler degrees at atmospheric pressure, and E_p - the viscosity sought for at pressure p. The following results were obtained in mineral oil tests at 40°C:

Pressure in kg/cm^2	0	157	315	472	630	787	945	1,100	1,300
Conditional viscosity in °E ..	3.24	4.53	6.48	9.71	14.45	21.33	28.8	41.65	60.0

The viscosity of vegetable oils changes together with pressure about twice as little as that of mineral oils. All drop fluids solidify under very high pressure; mineral oils of medium viscosity turn solid at 15 - 20 thousand kg/cm^2. The changing viscosity may fully compensate the fluid leakage in design spaces. There are 9 figures and 4 references: 1 Soviet and 3 English.

Card 3/5

88353

Figure 6: Schematic diagram of 7-cylinder pump for 280 kg/cm^2 pressure (self-explanatory)

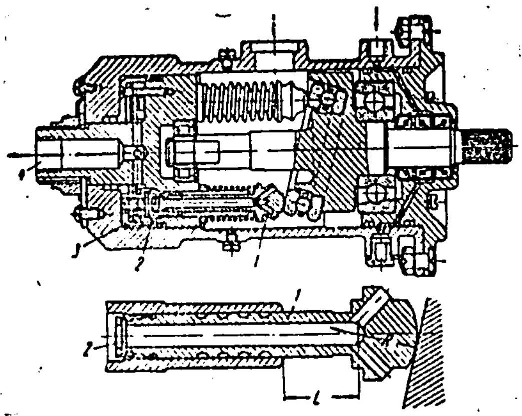
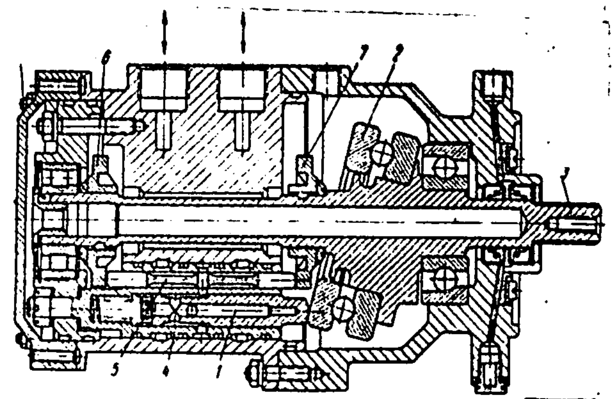


Figure 7: Schematic diagram of 5-cylinder pump for 280 kg/cm^2 pressure (self-explanatory)

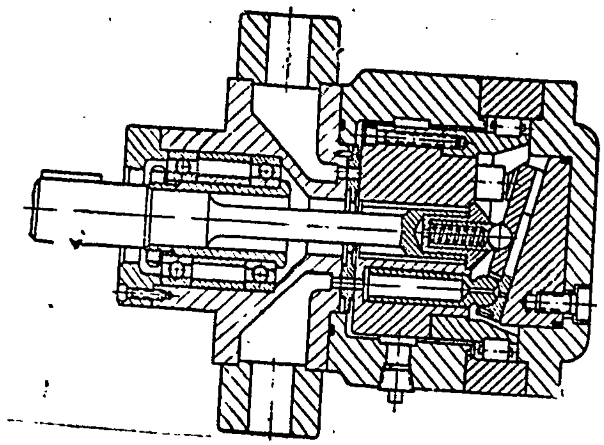


Card 4/5

Problems of High Pressure in Hydraulic Systems

88353
S/122/60/000/009/002/015
A161/A026

Figure 8: Schematic diagram of hydraulic engine for 350 kg/cm^2 pressure (self-explanatory)



Card 5/5

S/122/60/000/012/001/018
A161/A130

AUTHOR: Bashta, T. M., Professor Doctor of Technical Sciences

TITLE: Prolonging life of butt face distribution piston pumps

PERIODICAL: Vestnik mashinostroyeniya, no. 12, 1960, 3 - 10

TEXT: The article presents a discussion of design and operation of the distribution system of heavy duty hydraulic drive pumps with fluid distribution through the butt face (Fig. 1), consisting of a stationary distribution disc (1) and a rotary cylinder drum (2) that is pressed to the disc by a spring and the work fluid pressure in the cylinders. Such distribution units exist in flat (Fig. 1) and spherical (Fig. 4) type, and are used for pumps up to 4,000 hp producing up to 500 kg/cm^2 work pressure on the distribution disc. Their wear is the major factor determining the life of the drive pump. The material of the disc is cast iron, the drum is of nitrated hard steel, and the pistons of berillium bronze. The combination of nitrated steel disc with tin-lead bronze drum is also used. In small pumps the disc is sometimes made of graphite and the drum of steel. A diagram (Fig. 2) shows the windows in disc and the pressure distribution in the gap between the drum and the disc. The spherical type is recommended to prefer, for the possible slight

Card 1/5

tilt in it compensates manufacturing inaccuracies. The role of hair-thin grooves (4) is to connect the cylinders with the pressure space preliminarily to matching of a cylinder with the window in the disc, and so to prevent back surge. The pump capacity is adjusted and feed reversed by changing the α angle (Fig. 1). The effect of the oil film in the gap between the drum and the disc is analyzed with reference to investigation results in other analogously working systems where it had been stated that not hydrodynamical laws but intermolecular bond forces are acting [Ref. 1: T. M. Bashta, Samoletnyye gidravlicheskiye privody i ustroystva (Aircraft hydraulic drives and devices), Oborongiz, 1951; Ref. 2: T. M. Bashta, Voprosy konstruyirovaniya uplotneniy tortsovogo tipa, Stanki i instrument, no. 7, 1958]. It had been proven, that lubricant molecules are firmly bonded to metal on the boundaries and between themselves, that shearing stress is present in the boundary layers of such polar molecules preliminary to beginning of slip, and that stress varies from considerable at the metal surfaces to weaker in the gap mid. The lubricant film depth varied in experiments between 2 and 8 micron (depending on contact pressure, viscosity, etc.), and it was obvious that it is possible to stop practically all leak through the butt gap and to prolong work life of connections to many thousand hours. Too smooth surface finish is not advised, for slight roughness retains oil, and 6 - 8 micron roughness is considered permissible. It is recommended to reduce the sliding speed and the pressure in contact as far as possible; to prevent the formation of an oil wedge in the gap tending to separate the parts; to consider that the angular velocity of the drum can increase leak due to inertia when the drum butt face is not accurately perpendicular to the axis. It had been observed in experiments that oil was losing the lubricating capacity with time when leak was completely absent, and that periodical injections of small quantities of fresh oil had beneficial effect. The design in Fig. 4 includes such lubrication - one of the two holes (3) is connected to the suction space, and the other to the pressure space. Apart from this, the disc is provided with many shallow blind holes (4) joined with a narrow ring groove, and the drum butt face of the seven-cylinder drum with seven blind holes (6). The effect is seven short oil injections into the holes (4) during every revolution, and seven relieves, i.e., a pulsating oil cushion that considerably reduces wear without increasing leak. Some pressure relief and lubrication improvement is possible by using blind holes only, without forced oil feed, for the holes will produce a hydraulic cushion. There are 5 figures and 5 Soviet-bloc references.

PHASE I BOOK EXPLOITATION SOV/5815

Bashta, Trifon Maksimovich

Raschety i konstruktsii samoletnykh gidravlicheskikh ustroystv (Design and Construction of Aircraft Hydraulic Equipment) 3rd ed., rev. and enl. Moscow, Oborongiz, 1961. 476 p. 10,150 copies printed. Errata slip inserted.

Reviewer: I. I. Kukolevskiy (deceased), Doctor of Technical Sciences, Professor; Scientific Ed.: S. N. Rozhdestvenskiy, Candidate of Technical Sciences; Ed. of Publishing House: P. B. Morozova; Tech. Ed.: V. P. Rozhin; Managing Ed.: S. D. Krasil'nikov, Engineer.

PURPOSE: This book is intended for engineers and technicians in industry. It may also be useful to students at schools of higher technical education.

COVERAGE: The book describes various designs of aircraft hydraulic units and devices, presents design methods, and discusses the properties of hydraulic liquids. Attention is given to factors affecting the operation of hydraulic machines and the flow of liquid in hydraulic systems. The following topics are treated:

Card 1/2

hydraulic impulse, hydraulic theory of pipes, hydraulic motors and pumps (piston, vane, and gear types), rotary hydraulic drives, actuators, distributing systems, protective devices, remote control and servo systems, and packing devices. No personalities are mentioned. There are 17 references: 8 Soviet (1 translation), 8 English, and 1 German.

TABLE OF CONTENTS:

20605

S/147/61/000/001/014/016
E022/E135

26.2195

AUTHORS: Bashta, T.M., and Nikitin, G.A.

TITLE: The Influence of the Oscillatory Motion of the Piston on the Friction Force in Hydraulic Systems With Piston Valves

PERIODICAL: Izvestiya vysshikh uchebnykh zavedeniy, Aviatsionnaya tekhnika, 1961, No. 1, pp. 121-125

TEXT: Experiments with hydraulic piston valves employed in governing mechanisms etc. show that if the hydraulic pressure is large the frictional forces against the motion of the valve may increase up to a hundred times the corresponding values under low pressures (Refs. 1, 2). Such a large increase in resistance to the movement of the valve may cause partial or total failure of the system. In order to diminish the resistance, it is usual to make one element of the hydraulic pair either to oscillate with high frequency and small amplitude or to rotate about its axis. The earlier work of the author T.M. Bashta shows that such a rotational motion of the tube diminishes the force of resistance from 5 kg to 70 grams. The present work was carried out in order

Card 1/ 6

20605

S/147/61/000/001/014/016
E022/E135

The Influence of the Oscillatory Motion of the Piston on the Friction Force in Hydraulic Systems With Piston Valves

to evaluate the effect of the oscillatory motion of the piston on the resistance to its motion. The range of oscillations tested was from 0.1 to 2.4 mm, with the frequency of oscillations from 2 to 30 c.p.s. Oscillations of the piston were produced by an electromagnet which was connected axially with the piston and was fed by d.c. of alternate polarity using a rotary switch. This was driven by a variable speed (120 to 1800 r.p.m.) motor giving switching frequencies of 2 to 30 per sec. The piston valves tested were of two-collar type with a single groove between the collars; diameter of the collars 12 mm, width of the collars 10 mm. The shape of the valves was so chosen that under pressures used, they seized, as described in earlier work of the authors (same journal, 1959, No. 2). Experiments showed that the change of amplitude of oscillation in the range from 0.1 to 2.4 mm at the given pressures did not affect the magnitude of the seizing force, but the higher the pressure of the fluid the higher the value of the seizing force. As the frequency of oscillation increases, the
Card 2/6

20605

S/147/61/000/001/014/016
E022/E135

The Influence of the Oscillatory Motion of the Piston on the Friction Force in Hydraulic Systems With Piston Valves

seizing force diminishes at first but then becomes sensibly constant as shown in Fig.3. In order to determine the force necessary to move the valve under various pressures when the valve oscillated, a series of experiments was carried out up to a pressure of 400 kg/cm^2. The valve was kept under a given pressure for a minute and then it was moved. Fig.4 shows the results of these experiments for three different valves, a, 6, and B. Ringed dots indicate the forces necessary to shift the valve without oscillations, while solid dots indicate the corresponding forces with the same valves when oscillating. It is clearly seen that oscillations reduce the resistance to the motion of the valve. Finally, some experiments were carried out to find out the effect of a sudden drop of the pressure in the system. Figs. 5 and 6 show these results. The pressure was built up to 500 kg/cm^2, kept at that value up to 20 minutes and then suddenly reduced. As is seen from Fig.5, the forces necessary to move the piston in those circumstances fall
Card 3/4

20605

S/147/61/000/001/014/016
E022/E135

The Influence of the Oscillatory Motion of the Piston on the Friction Force in Hydraulic Systems With Piston Valves

substantially when the piston oscillates (ring dots) as compared with those when it does not (solid dots). The explanation is that when the piston is not oscillating the flow of the fluid past it is greatly reduced with time at high pressures, while with the piston oscillating the flow of the fluid is not impaired at all, as shown in Fig.6.
There are 6 figures and 3 Soviet references.

ASSOCIATION: Kiyevskiy institut GVF, Kafedra aeromekhaniki i gidravliki
(Chair of Aeromechanics and Hydraulics, Kiyev Institute of the GVF)

SUBMITTED: May 3, 1960

Card 4/6

S/102/61/000/002/005/005
D251/D302

16,8000

AUTHOR: Bashta, T.M.

TITLE: The problem of using high liquid pressures in automatic systems

PERIODICAL: Avtomatyka, no. 2, 1961, 90 - 93

TEXT: The author considers certain properties of high liquid pressures which have a bearing on the use of such pressures in automatic systems. Graphs are given of the variation of mass of the hydrosystem against pressure, force of static friction of the plunger against pressure, and the number of cycles of load of the tube conductor of pulsating pressure on the ovalform of cross-section. It is stated that liquids with high chemical stability at high pressures have also high mechanical stability at these temperatures. The effect of high pressures on the accuracy and stability of tracing hydraulic systems is considered, and it is stated that increase of pressure increases the speed and accuracy, but decreases the resistance to auto-oscillation. It is shown that undissolved air in a

Card 1/2

mechanical solution worsens the conditions of operation of the hydro-system. The effect of high pressures on the use of a pump is also considered. There are 3 figures.

APPROVED FOR RELEASE: 06/06/2000 CIA-RDP86-00513R000203820019-6"

32014
S/569/61/005/000/001/002
D221/D302

11.0910

AUTHOR: Bashta, T.M.

TITLE: The problems of applying high pressures of liquid in automatic systems

SOURCE: International Federation of Automatic Control, 1st Congress, Moscow, 1960. Avtomatizatsiya proizvodstvennykh protsessov; mashinostroyeniye, elektroenergetika, elektroprivod, transport. Izd-vo AN SSSR, 1961. (Its: Trudy [t.5]), 38-49

TEXT: The advantage of weight decrease due to application of high pressure in hydraulic systems is lost at a certain point, when the design becomes heavier owing to strength requirements. The risk of fire is reduced with pressures higher than 280 kg/cm^2. The lower weight of high pressure systems reduces the inertia of moving parts and ensures a faster response. The improved volumetric efficiency and greater modulus of elasticity of the fluid provide a stable operation at low output

Card 1/4

23014
S/569/61/005/000/001/002
The problems of applying high pressures D221/D302

speeds. These units are eminently suitable for tracer controls. In distribution valves the friction increases with higher pressures up to a point, and then drops practically to 0 with a further growth of the latter. This paradox is explained by deformation of the parts of the valve in consequence of pressure. Experimental investigations showed that the electric resistance of the oil layer between the parts also varies with pressure. New problems appeared, especially that of securing the fatigue strength of pipes subject to pulsating high pressure. The risk of failure is especially great when the circular shape of the cross-section of the pipe is distorted. In certain cases there are resonant vibrations of the pipeline; the most probable and most destructive ones are the bending vibrations. Mineral liquids are capable of changing their viscous and lubricating properties when subject to prolonged mechanical action under high pressure, and their chemical properties change with temperature. The extension of temperature range of operation requires application of special liquids, of which silicon liquids are the most promising. They dissolve all existing plasticizers of synthetic rubbers, and exhibit unsatisfactory lubrication indices, and high

Card 2/4

fluidity. They are good, however, in the case of operation with hard steel - soft bronze pairs. High pressure increases the amplification factor of a tracer control system but reduces its resistance to self-oscillations (hunting). Insufficient rigidity of pipes and the elasticity of the fluid decrease the accuracy of tracer systems and may cause a loss of stability. Appropriate design of control valve can increase the stability to some degree. Stepped orifices are mentioned. The volumetric efficiency of high pressure pumps is

$\eta_{vol} = 1 - p_n\beta \frac{q_d}{q_p} - p_n\alpha \frac{q_d}{q_p} = 1 - p_n \cdot \frac{q_d}{q_d} (\beta + \alpha)$, where p_n is the pressure to which the fluid is compressed, q_d the dead volume above the pistons, q_p the volume swept by the piston during one stroke, the mean value of compressibility when pressure is increased by 1 atm (i.e. the reciprocal value of the modulus of elasticity of the liquid) and α a coefficient characterizing the change of dead volume during a compression of 1 atm. By the introduction of $r = \frac{q_p + q_d}{q_p}$ the author deduces

32014
S/569/61/005/000/001/002
The problems of applying high ... D221/D302

$\eta_{vol} = 1 - p_n(r-1)(\beta + d)$. The parameters r and α must be brought to minimum in the design of pumps. The volumetric coefficient of efficiency changes if the stroke of piston is varied for regulating the delivery of the pump. The presence of undissolved air in the liquid is very undesirable. The change of viscosity of mineral oils is given by the empirical equation $\nu_p = (1 + 0.003p)\nu$, where ν is the dynamic viscosity at atmospheric pressure, and ν_p is the viscosity at pressure p. All drop-forming liquids solidify at sufficiently high pressures, e.g. mineral oils of medium viscosity at 15-20 thousand atm. The change of viscosity must be taken into account in calculating leakage; this factor can fully compensate the expected increase of leakages, due to the widening of constructional gaps under high pressure. There are 6 figures and 1 table.

Card 4/4

28159

18.8300 S/122/61/000/009/001/009
D298/D305

AUTHOR: Bashta, T.M., Doctor of Technical Sciences, Professor

TITLE: Liquid cavitation in hydraulic installations

PERIODICAL: Vestnik mashinostroyeniya, no. 9, 1961, 7-12

TEXT: This paper discusses the process of cavitation - discharge of liquid vapors which subsequently condense. As a result of this process, local hydraulic hammers appear and cause a sharp rise in temperature and pressure in the centers of condensation. During a lasting cavitation, surface destruction (erosion) of hydraulic installation components occurs in places where the vapor bubbles condense. Cavitation can appear in pipe-lines, pumps and other devices where the liquid flux undergoes narrowing with a subsequent expansion, e.g. in cocks, valves, diaphragms, etc. The author describes in detail the

X

Card 1/3

28159

Liquid cavitation in ... S/122/61/000/009/001/009
D298/D305

process of cavitation and gives pertinent graphs and layouts. At the same time he shows the methods for suppressing cavitation and its consequences. For this purpose he recommends using corrosion-resisting materials (steel containing chrome and nickel) for making the components of hydraulic installations. Simultaneously, the surface of these components must be thoroughly finished. With the increase of machining quality, stability of components against cavitational destruction is, as a rule, increased. In practice it has been established that using non-corrosive steel, brands НВ150(NV-150) to НV400-420 (NV400-420), makes the components approximately 10 times more resistive to cavitation. For this purpose carbon steel can also be used. Satisfactory anti-cavitational stability may be found in hard bronze, while cast iron proved to be unsatisfactory. Another factor contributing to decrease of cavitation is the buoyancy of vapor in hydraulic liquid. For this reason, a two-component mixture АМГ-10(AMG-10), containing kerosene as

Card 2/3

28159

Liquid cavitation in ... S/122/61/000/009/001/009
D298/D305

a component, is widely used for hydraulic installations. There are 10 figures and 4 references: 2 Soviet-bloc and 2 non-Soviet-bloc. The reference to the English-language publications reads as follows:

Hydraulic Power Transmission, 1959,5,No 52.
Product Engineering, 1959, 30, No 9.

Card 3/3

BOGDANOVICH, Leonid Boleslavovich; BASHTA, T.M., doktor tekhn. nauk, prof., retsenzent; GORELKIN, A.V., kand. tekhn. nauk, dots., red.; RIKBERG, D.B., red.; GORNOSTAYPOL'SKAYA, M.S., tekhn. red.

[Hydraulic drives in machinery; diagrams and designs]Gidravlicheskie privody v mashinakh; skhemy i konstruktsii. Moskva, Mashgiz, 1962. 222 p. (MIRA 16:3)
(Machinery--Hydraulic drive)

37132
S/122/62/000/004/004/006
D221/D302

26.2190

AUTHOR: Bashta, T.M., Doctor of Technical Sciences, Professor

TITLE: The metering of slow fluid flows

PERIODICAL: Vestnik mashinostroyeniya, no. 4, 1962, 43 - 49

TEXT: The article describes throttle and volumetric types of controlling valves. The former is represented by a multidiaphragm unit. When operating under a low pressure difference, there are limitations on the size of holes in the washers due to risk of clogging. The difficulty may be obviated by providing a relative translatory and rotary motion to the diaphragms. The stability of operation of the valve is also related to its position. When placed in the return circuit, they allow control of double acting cylinders, and at the same time reveal anti-oscillatory characteristics. The superiority of return branch position over the mounting in the supply line is especially marked in the case of air inclusion in the fluid. In addition, the heat due to throttling is conveyed to the sump. The effect of throttling on the speed of the hydraulic motor is reduced by the incorporation of a pressure regulator whose resistance is al-

Card 1/2

ways equal to the difference between the pressure at the inlet and the constant pressure set by the regulator. Volumetric metering ensures a stability of flow within the range of 20 - 30 cm^3/min. In its simplified form the meters constitute of a variable delivery pump, usually with a controlled working volume which possesses a high volumetric efficiency coefficient. The output is regulated by an inclined swash plate. As before it is advantageous to mount the regulator in the return line. The minimum flow depends on the amount of leakage in the system. In actual units, it is related to the pressure drop in the regulator, which in turn depends on the load of the hydraulic motor. The control of flow by change of pump speed is inferior to the above described method. The uniformity of control of small deliveries is enhanced by introduction of differential volumetric regulators. A detailed examination of the arrangement is given, and the corresponding equations are derived. They demonstrate the effect of leakages on the actual delivery as well as on the volumetric coefficient of the unit. This may be minimized by the control of pressure drop in the pumps. An illustration of the suggested method is quoted. There are 9 figures and 4 Soviet-bloc references.

BASHTA, T.M.; ZAYCHENKO, I.Z., doktor tekhn. nauk, retsenzent; SOKOLOVA, T.F., tekhn. red.

[Hydraulics in the manufacture of machinery] Mashinostroitel'naia gidravlika; spravochnoe posobie. Moskva, Mashgiz, 1963. 696 p. (MIRA 17:1)

AM4024707 BOOK EXPLOITATION S/

Bashta, T. M.

Hydraulic systems in machine building; a reference manual (Mashinostroitel'naya gidravlika; spravochnoye posobiye). Moscow, Mashgiz, 63. 0696 p. illus., bilio., index. Errata slip inserted. 23,00 copies printed.

TOPIC TAGS: hydraulic system, hydraulic control, hydraulic control equipment, hydraulic motor, hydraulic amplifier, valve gear, piping, pump, hydraulic converter, fluid flow, fluid filter, packing material, stuffing material, hydraulic equipment manufacture, hydraulic equipment tests

PURPOSE AND COVERAGE: The book presents data on the selection, calculation, construction, manufacture, and use of three-dimensional hydraulic systems in different branches of machine building. The topics include: design of pumps; hydraulic motors; power cylinders; hydraulic transmissions; distribution; protection, and regulating equipment; hydraulic servomechanisms; and packing and filtering units;

Card 1/3

AM4024707

Detailed data are presented on a large number of working fluids along with recommendations for their use. Formulas and tables are presented to simplify the design of hydraulic systems. The technical specifications for materials used in the manufacture of hydraulic units, for the accuracy and surface-finish tolerances, and for tests of hydraulic systems are presented in detail. The book is intended for engineering-technical workers in machine design.

TABLE OF CONTENTS [abridged]:

AM4024707

SUB CODE: MD, MA, FL SUBMITTED: 29Aug63 NR REF SOV: 166

OTHER: 013 DATE ACQ: 12Mar64

Card 3/3

APPROVED FOR RELEASE: 06/06/2000 CIA-RDP86-00513R000203820019-6"

Card 1/2

Card 2/4

L 35932-66 EWT(m)/T DJ

ACC NR: AP6014333 (A, N) SOURCE CODE: UR/0122/65/000/012/0023/0028

56
B

AUTHOR: Bashta, T. M. (Meritorious in science and technology, Doctor of technical sciences, Professor)

ORG: None

TITLE: Fluid hammer in the hydraulic systems of machines

SOURCE: Vestnik mashinostroyeniya, no. 12, 1965, 23-28

TOPIC TAGS: fluid pressure, fluid velocity, hydraulic device, shock wave, shock wave oscillation, VIBRATION

ABSTRACT: The author considers the phenomenon of fluid hammer (fluid pressure vibrations) which takes place when a hydraulic mechanism is suddenly activated or stopped. The basic relationships are given for determining the increase in pressure which accompanies the shock in terms of the fluid density, speed of the shock wave and loss of fluid velocity. It is shown that flexible connections may be used to reduce the amplitude and duration of vibrations resulting from fluid hammer. Special tests showed that maximum pressures of 4 times the working pressure are generated when the piston in a power cylinder is suddenly stopped. The amplitude of the fluid hammer may be reduced by using choke type time delay relays to increase the distributer connection time, or various types of shock compensators (dampers) to reduce the period of the

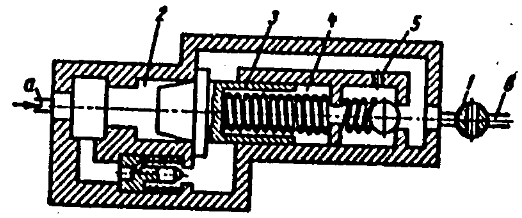
Card 1/2

UDC: 621.22.018.8

L 38682-66

ACC NR: AP6014333

pipe (the time required for the shock wave to travel from the valve to the source of flow and back again). One type of choke relay is shown in the figure. When valve *1* is opened, pipelines *a* and *b* are connected at first only through the narrow aperture *2*, and only when valve *3* slides a predetermined distance to the right, pushing the fluid from cavity *4* through choke orifice *5* is the valve completely opened.



The shock compensators are usually a vessel (reservoir) with some type of elastic element. The most widely used compensators employ pistons with spring or gas elements for elasticity. The shock pressure is reduced by absorption of part of the energy of the shock wave in deformation of the elastic element in the compressor. The use of safety valves for limiting shock pressure is also discussed. Orig. art. has: 10 figures.

SUB CODE: 13/ SUBM DATE: none/ ORIG REF: 000/ OTH REF: 000

Card 2/2 vmb

ACC NR: AP6019189 (N) SOURCE CODE: UR/0122/66/000/002/0020/0024

AUTHOR: Bashta, T. M. (Honored scientist and technician, Doctor of technical sciences, Professor)

37
B

ORG: None

TITLE: Problems in sealing the output shafts of hydraulic generators

SOURCE: Vestnik mashinostroyeniya, no. 2, 1966, 20-24

TOPIC TAGS: hydraulic engineering, sealing device, rotating seal, hermetic seal

ABSTRACT: The author considers problems involved in hermetically sealing rotating members for operation at high speed under high-pressure conditions. It is pointed out that failure of rubber seals is due to hardening of the rubber after protracted action of high temperatures. This condition is accelerated by additives in the oils used for increasing their lubricating properties. The surface of the shaft in contact with the sealing sleeve should have at least a 10th class finish, but should not be too smooth since excessive smoothness increases friction and wear. Tests show that irregularities of 0.5 μ on the shaft surface give minimum friction in rubber sealing sleeves. Wear resistance of the shaft may be increased by chrome plating. Sealing rings operate more effectively if they are set at an angle to the shaft axis. Rings set at an angle of 3° with the plane perpendicular to the shaft show a reduction in the coefficient of fric-

Card 1/2 UDC: 621.225-762.6 621.65-762.6

L 41218-66

ACC NR: AP6019189

0

tion by a factor of 2. Setting the rings at an angle also increases the contact surface and acts as a better heat sink. An increase in the angle of inclination of the sealing ring improves operating conditions at higher hydraulic pressures. However, an increase in this angle is accompanied by complications in making the groove and also by an increase in leakage. An angle of 3.5-4° is recommended. The cross section of the ring and the depth of the groove should be selected to give a radial compression of 8-12%. Recommendations are given for the use of lubricants with respect to operating conditions and shaft dimensions. Orig. art. has: 10 figures.

SUB CODE: 13/ SUBM DATE: none

Card 2/2 mLP

APPROVED FOR RELEASE: 06/06/2000 CIA-RDP86-00513R000203820019-6"

BASHTAN, F.A.

CA

1

An electric laboratory still. F. A. Bashtan and L. I. Chernetskii. *Lab. Prakt.* (U. S. S. R.) 16, No. 12, 23-4 (1941).—A compact glass app. for the production of small quantities of distd. water is provided with heating electrodes of rustproof sheet steel in the form of 2 concentric tubes (outer tube 200 mm. long, diam. 25 mm.; inner tube 200 mm. long, diam. 21 mm.) insulated from each other by 6 porcelain insulators and mounted on an asbestos-cement panel. The whole app. is mounted in a wooden box. The condenser is connected with a water tap which regulates the flow of water. The capacity of the still is approx. 900 ml./hr. The app. is automatic. The scale accumulating on the electrodes and the walls of the app. is removed with 10-20% AcOH, poured through the condenser. The time required for this is 5-10 min. The app. can be used for an indefinite period of time. W. R. Henn

ASM-SLA METALLURGICAL LITERATURE CLASSIFICATION

BASHTAN, F. A.

22002 BASHTAN, F. A. K voprosu ob opredelenii mysh'yaka. Vracheb. delo, 1949, No. 7, stb. 629-32.

SO: Letopis' Zhurnal'nykh Statey, No. 29, Moskva, 1949.

BASHTAN, F.A.; GOLOBIN, D.I.; STAPANOVA; SHUL'MAN, A.A.

Prevention of Basedow's disease caused by iodine. Vrach.delo no.2:201 F '57. (MLRA 10:6)

1. Chernovitskiy meditsinskiy institut.
(POTASSIUM IODIDE--TOXICOLOGY)
(GRAVES' DISEASES)

BASHTAN, N. S.

BASHTAN, N. S. -- "Investigation of the Moisture Cycle in the Forest Zone of the European Portion of the Soviet Union." Leningrad Order of Lenin State U imeni A. A. Zhdanov. Leningrad, 1955. (Dissertation for the Degree of Candidate in Geographical Sciences.)

SO: Knizhnaya Letopis', No 5, Moscow, Feb 1956

BASHTAN, N.S.

Water economy in the forest zone of the European part of the U.S.S.R. Uch.sap.LGU no.269:50-64 '59. (MIRA 12:6)
(Water)

BASHTAN, N.S.

Processes resulting in precipitation in the forest zone of the European part of the Soviet Union. Uch.zap.LGU no.269:141-159 '59. (MIRA 12:6)
(Precipitation (Meteorology))

DROZDOV, Oleg Alekseyevich, doktor geogr. nauk; GRIGOR'YEVA, Anna Sergeyevna, kand. geogr. nauk. Prinimal uchastiye BASHTAN, N.S., assistent; POKROVSKAYA, T.V., otv. red.; [redacted], A.B., red.; BRAYNINA, M.I., tekhn. red.

[Moisture circulation in the atmosphere] Vlagooborot v atmosfere. Leningrad, Gidrometeoizdat, 1963. 314 p. (MIRA 16:8)

1. Kafedra meteorologii geograficheskogo fakul'teta Leningradskogo gosudarstvennogo universiteta (for Bashtan). (Moisture)

BASHTAN, N.S.

Moisture circulation and precipitation forecasting. Vest. LGU 20 no.6:108-114 '65. (MIRA 18:4)

BASHTAN, N.S.

Transformation of air masses. Mat.po meteor.i klim. no.1:9-19 '63.

Air parcels traveling from the eastern area of Asia to the north of the European part of the U.S.S.R. Ibid.:20-23 (MIRA 17:3)

BASHTANNIK, Kirill Georgiyevich [deceased]; D'YAKONOV, V.F., nauchnyy redaktor; SAVCHENKO, K.N., nauchnyy redaktor; IVANOV, K.A., redaktor izdatel'stva; TIKHONOVA, Ye.A., tekhnicheskiy redaktor

[Nautical astronomy] Morekhodnaia astronomiia. Moskva, Izd-vo "Morskoi transport," 1956. 318 p. (MIRA 10:4)
(Nautical astronomy)

BASHTOVOY, I.A.; GRITSENKO, A.M.; KUCHERENKO, S.K.; MIKHAYLENKO, F.K.; SELYUTIN, I.A.

Drawing rock pillars in deepening mine shafts.
Sbor.rats.predl.vnedr.v proizv. no.1:5-6 '61. (MIRA 14:7)

1. Trest "Dzerzhinskruda", rudnik im. Kirova.
(Mining engineering)

BASHUK, I. A.

KRAFT, M. Ya., and BASHUK, I. A.

Mbr., Inst. Pharmacology, Chemical Therapy, and Toxicoloty, Moscow, -1946-.

"Reaction of Arseno-Compounds with Arsenical Acids and Oxides of Arsenic," Dok. AN, 55, No. 5, 1947

"Reaction of 3,3'-Diamino-4,4'-Dioxyarsenobenzene with Sulfuric Acid," Dok. AN, 55, No. 8, 1947

105-6-24/26

AUTHOR BASHUK, I.B., Candidate of Techn. Sciences, Docent

TITLE Properties and Characteristics of Large Germanium and Silicon Rectifiers. (Svoystva i kharakteristiki moshchnykh germaniyevykh i kremniyevykh vypryamiteley - Russian)

PERIODICAL Elektrichestvo, 1957, Nr 6, pp 91 - 93 (U.S.S.R.)

ABSTRACT In the USSR germanium rectifiers are being built up to 350 and in other countries up to 200 A. This papers deals with the construction of a germanium rectifier with water cooling and a rectifier element of 200 A, 65 V D.C. From the diagrams shown here it may be seen that the most important property of germanium rectifiers is the low voltage drop in the conductive part of the period itself at high current densities. In comparison to selenium rectifiers losses in this case are negligibly small. Germanium rectifiers with natural and air cooling are 40 times smaller than the corresponding selenium rectifiers. When determining the boundary values of rectifiers it is necessary to take the influence exercised by temperature upon the volt-ampere characteristic into account both in the conductive and in the nonconductive direction. The possibilites of improving the parameters of these rectifiers have as yet not been exhausted. Apart from germanium, also silicon can be used with success for large rectifiers. Its essential advantage is its ability of being able to operate at high temperatures of up to 300°C. Its disadvantages are- a voltage drop that is more straight than in the case of germanium, and the difficulty of cleaning. One of the 10kW silicon rectifier operates at 200°C. IN connection with the rectification of a three-phase current according to a bridge scheme such a rectifier produces a rectified cur-

AS
PR
SU
AV.
Ca

Card 1/2

AUTHOR: 1) VEYTS, V.I., Corresponding member of Academy of Science of the U.S.S.R. 105-8-18/20
2) Author not given
3) BASHUK, I.B., Ass.Prof., cand.techn.sc.
4) GALONEN, Yu.M., cand.techn.sc.

TITLE: 1) Refers to the Article by John HARDT. (Po povodu stat'i Dzhona Khardta, Russian)
2) An American Magazine on the Soviet Power Economy. (Amerikanskiy zhurnal o sovetskoy energetike, Russian)
3) On the Industrial Use of Strong Germanium Rectifiers. (Promyshlennoye primeneniye moshchnykh germaniyevykh vypryamiteley, Russian)
4) The Urban Railless Electric Traffic Abroad. (Gorodskoy bezrel'sovyy elektrotransport za rubezhom, Russian)

PERIODICAL: Elektrichestvo, 1957, Nr 8, pp 77 - 90 (U.S.S.R.)

ABSTRACT: 1) A criticism of and answer to the article in "Electrical Engineering", Vol 75, p 978, Nr 11, 1956. The tendentious character of the article is deplored, a number of other, unbiassed English publications are pointed out and a survey on the present state of development in the U.S.S.R. is given. (12 Slavic references)

Card 1/2

2) An American Magazine on the Soviet Power Economy. 105-8-18/20
3) On the Industrial Use of Strong Germanium Rectifiers.
4) The Urban Railless Electric Traffic Abroad.

2) Concerns the article by Andrew W.KRAMER and Richard H.MORRIS in "Power Engineering", 1956, October, p 78 - 81
3) A survey based on the papers by C.F.MACHIN in "Research", 1955, Vol 8, Nr 7, p 262, R.M.CRENSHOW in El.Eng., 1955, Vol 74, Nr 5, p 418 and Vol 75, 1956, Nr 8, p 719, The Engineer, 1956, Vol 201, Nr 5216, p 53, and "Modern Transport", 1956, March 24.
4) A survey of the Development of filo busses, accumulator electromobiles, electro-giro busses and of the projects and tests of railless express lines outside the streets.

ASSOCIATION: Not given
PRESENTED BY:
SUBMITTED:
AVAILABLE: Library of Congress

Card 2/2

32(3)

SOV/112-59-5-9121

Translation from: Referativnyy zhurnal. Elektrotekhnika, 1959, Nr 5, p 102 (USSR)

AUTHOR: Bashuk, I. B., and Zorokhovich, A. Ye.

TITLE: Using Semiconductor Rectifiers on Diesel-Electric Locomotives

PERIODICAL: Elektr. i teplovozn. tyaga, 1957, Nr 12, pp 36-37

ABSTRACT: With 3,000-5,000 hp gas turbines and diesels used on gas-turbine locomotives and diesel-electric locomotives, difficulties arise in creating a simple and reliable system of electrical transmission. These difficulties are due to the fact that a DC generator cannot be designed for such capacity because of cooling, commutation, and mechanical strength conditions which all depend on its speed. Modern generator designs go as high as 1,500 kw (with permissible loading of 500 amp per 1 cm of armature circumference, average bar-to-bar voltage 28 v and peripheral armature speed 50 m/sec). With higher-speed prime movers, the generator capacity becomes lower. These facts require that a number of generators be coupled to the gas-turbine shaft,

SOV/112-59-5-9121

Using Semiconductor Rectifiers on Diesel-Electric Locomotives

which results in complicating both power plant and electrical schemes. Use of reducers appreciably increases the weight and size of the generator apparatus. The problem of increasing both capacity and speed of the generator can be successfully solved by using 3-phase synchronous generators and semiconductor rectifiers connected in a 3-phase bridge circuit. Rectified-current ripple amounts to only 6%, so that smoothing reactors are not needed. In this scheme, the synchronous-generator field winding is fed from a synchronous exciter also via semiconductor rectifiers. The field is controlled by an automatic regulator. This scheme retains conventional DC traction motors, only one generator is needed for a diesel-electric locomotive of any capacity, and the generator size decreases with increasing speed. A speed increase from 1,000 to 3,000 rpm permits cutting the generator length and diameter 1.32 times, and weight 2.28 times. The size of 3,000-4,000-kw silicon rectifiers is only 0.5-0.7 m^3. This type of transmission leads to simplified equipment,

Card 2/3

SOV/112-59-5-9121

Using Semiconductor Rectifiers on Diesel-Electric Locomotives

lesser maintenance, and increased reliability of the power plant. A synchronous-generator locomotive can also be used as a mobile electric generating station.

L.A.Ch.

Card 3/3

BENEDIKT, O.V.; BASHUK, I.B.; ZOROKHOVICH, A.Ye.; SAZONOV, inzh., red.; VERINA, G.P., tekhn.red.

[Electric drives in diesel- and gas-turbine locomotives equipped with a. c. machinery] Elektricheskie peredachi teplovozov i gazoturbovozov s mashinami peremennogo toka. Moskva, Gos. transp. zhel-dor. izd-vo 1958. 78 p. (Moscow. Moskovskii institut inzhenerov zheleznodorozhnogo transporta. Trudy, no.106). (MIRA 11:4)
(Locomotives--Electric driving)

BASHUK, I.B., dots, kand. tekhn. nauk.

Selecting electric transmission systems for electric and diesel locomotive speedometers. Trudy MIIT no.95:25-42 '58. (MIRA 11:12)
(Locomotives--Electric equipment) (Speedometers)

BASHUK, I.B., kand. tekhn. nauk; SKIBA, L.I., inzh.

Parallel operation of semiconductor valves. Trudy TSNII MPS no.170:179-198 '59. (MIRA 12:7)

(Valves)

BASHUK, I.B., kand.tekhn.nauk, dotsent; KHOMENKO, A.I., kand.tekhn. nauk, dotsent

Contactless control of electric railroad motorcars. Elektrichestvo no.10:24-28 O '61. (MIRA 14:10)

1. Moskovskiy institut inzhenerov zheleznodorozhnogo transporta. (Railroad motorcars)

BASHUK, I.B., kand.tekhn.nauk, dotsent; KHOMENKO, A.I., kand.tekhn.nauk, dotsent

Contactless valve switch for the ER1 electric train. Elek.i tepl. tiaga 5 no.10:9-10 O '61. (MIRA 14:10)

1. Moskovskiy institut inzhenerov zheleznodorozhnogo transporta.
(Railroad motorcars)

BASHUK, M.B.; SHAPIRO, A.S.

Determining the economic effectiveness of capital investments in industrial construction. Khim.prom. no.2:70-75 Mr '54. (MLRA 7:6)

1. Giprokhim. (Chemical plants--Cost of construction)

BASHUK, Petr Petrovich, brigadir kompleksnoy brigady kamenshchikov; RAZINKOV, P. redaktor; LIL'IN, A., tekhnicheskiy redaktor

[Combined work teams in construction work] Kompleksnaia brigada na stroitel'stve. Moskva, Moskovskii rabochii, 1956. 46 p. (MIRA 10:4)

(Construction industry)

82510

15.2210

S/070/60/005/004/012/012
E132/E360

AUTHORS: Bashuk, R.P., Basayev, V.P., Tsadkina, R.B. and ~~Fayvusovich,~~ S.A.

TITLE: The Hydrothermal Synthesis of Corundum with Impurities 15

PERIODICAL: Kristallografiya, 1960, Vol. 5, No. 4, pp.666-667

TEXT: The exploitation of paramagnetic materials based on the corundum structure demands the introduction into the lattice of paramagnetic ions. Cr can be introduced by the Verneuil process but not Fe, Ti nor other elements. Nevertheless, natural specimens exist with significant quantities of these impurities. The hydrothermal methods successfully used in the USSR for growing quartz can also be used for corundum. Specimens made in this way were tested radiospectroscopically and by X-ray methods. It has been shown that Fe^{+++} ions entered the lattice isomorphously replacing the Al^{+++} ions. The concentrations achieved were one or two orders of magnitude greater than could be achieved by the Verneuil process. Spherical seeds gave crystals with the following simple forms: $\{0001\}$, $\{10\bar{1}1\}$, $\{22\bar{4}3\}$ and $\{22\bar{4}1\}$. There are 2 figures and 2 references: 1 German and 1 English.

SUBMITTED: March 22, 1960

Card 1/1

S/070/60/005/004/011/012
E132/E360

AUTHORS: Bashuk, R.P., Vol'pert, E.G. and Tsigler, I.N.

TITLE: Annealing Boules of Synthetic Corundum 15

PERIODICAL: Kristallografiya, 1960, Vol..5, No. 4, p 643

TEXT: Crystals of synthetic corundum, grown by the Verneuil process, and known as boules, have a considerable residual strain as a result of which they usually split in half longitudinally under a slight blow or scratch. In most cases, the six-fold axis of the corundum coincides with the axis of the boule. Only half boules are normally used industrially and this limits the size of the objects which can be made. Trials of annealing as a method of removing such strains have been made and these have proved successful, enabling plate to be cut freely parallel and perpendicular to the optic axis of the crystal. There are 1 figure and 1 Soviet reference. √B

SUBMITTED: March 22, 1960

Card 1/1